

NOVEL INTEGRATED RADIAL AND AXIAL MAGNETIC BEARING

Kenneth A. Blumenstock

NASA's Goddard Space Flight Center, Code 544, Greenbelt, Maryland USA
kenneth.a.blumenstock.1@gsfc.nasa.gov

Gary L. Brown

NASA's Goddard Space Flight Center, Code 544, Greenbelt, Maryland USA
gary.l.brown.1@gsfc.nasa.gov

ABSTRACT

Typically, fully active magnetically suspended systems require one axial and two radial magnetic bearings. Combining radial and axial functions into a single device allows for more compact and elegant packaging. Furthermore, in the case of high-speed devices such as energy storage flywheels, it is beneficial to minimize shaft length to keep rotor mode frequencies as high as possible. Attempts have been made to combine radial and axial functionality, but with certain drawbacks. One approach requires magnetic control flux to flow through a bias magnet reducing control effectiveness, thus resulting in increased resistive losses. This approach also requires axial force producing magnetic flux to flow in a direction into the rotor laminate that is undesirable for minimizing eddy-current losses resulting in rotational losses. Another approach applies a conical rotor shape to what otherwise would be a radial heteropolar magnetic bearing configuration. However, positional non-linear effects are introduced with this scheme and the same windings are used for bias, radial, and axial control adding complexity to the controller and electronics. For this approach, the amount of axial capability must be limited.

It would be desirable for an integrated radial and axial magnetic bearing to have the following characteristics: separate inputs for radial and axial control for electronics and control simplicity, all magnetic control fluxes should only flow through their respective air gaps and should not flow through any bias magnets for minimal resistive losses, be of a homopolar design to minimize rotational losses, position related non-linear effects should be minimized, and dependant upon the design parameters, be able to achieve any radial/axial force or power ratio as desired. The integrated radial and axial magnetic bearing described in this paper exhibits all these characteristics. Magnetic circuit

design, design equations, and analysis results will be presented.

INTRODUCTION

An R&D project at NASA's GSFC called MASAREDI (Magnetically Suspended Actively Reduced Dynamic Imbalance) necessitated retrofitting magnetic bearings into an Ithaco™ size E momentum reaction wheel. The purpose was to demonstrate the jitter (vibration) performance improvement that automatic balance control can achieve as compared with a characterized flight wheel using conventional mechanical bearings. It was decided that it would be beneficial to combine radial and axial functions into a single magnetic bearing component in order to reduce the number of components and length of the shaft. Recent projects at NASA/GSFC used either radial force only homopolar magnetic bearings [1], or combined radial homopolar in tandem with a separate axial magnetic bearing [2].

Traditional methods of combining radial and axial functions were studied. Heteropolar magnetic bearings can be made conical and fluxes controlled so as to provide radial and axial control. In order to minimize eddy-current losses, it was deemed undesirable to go with a heteropolar configuration. Heteropolar designs also require continual parasitic power dissipation to provide bias flux if linear operation is desirable. In addition, a strong coupling effect between axial and radial axes is introduced which limits the amount of axial force capability that can be incorporated into the design. For this reason, only a small conical angle and only a small percentage of axial force capability are practical for a design to minimize the coupling effect. Since the momentum wheel was required to operate with its axis vertical in the lab, and the wheel mass was a substantial 12 kg, it was necessary to have high force capability with low power dissipation in the axial direction.

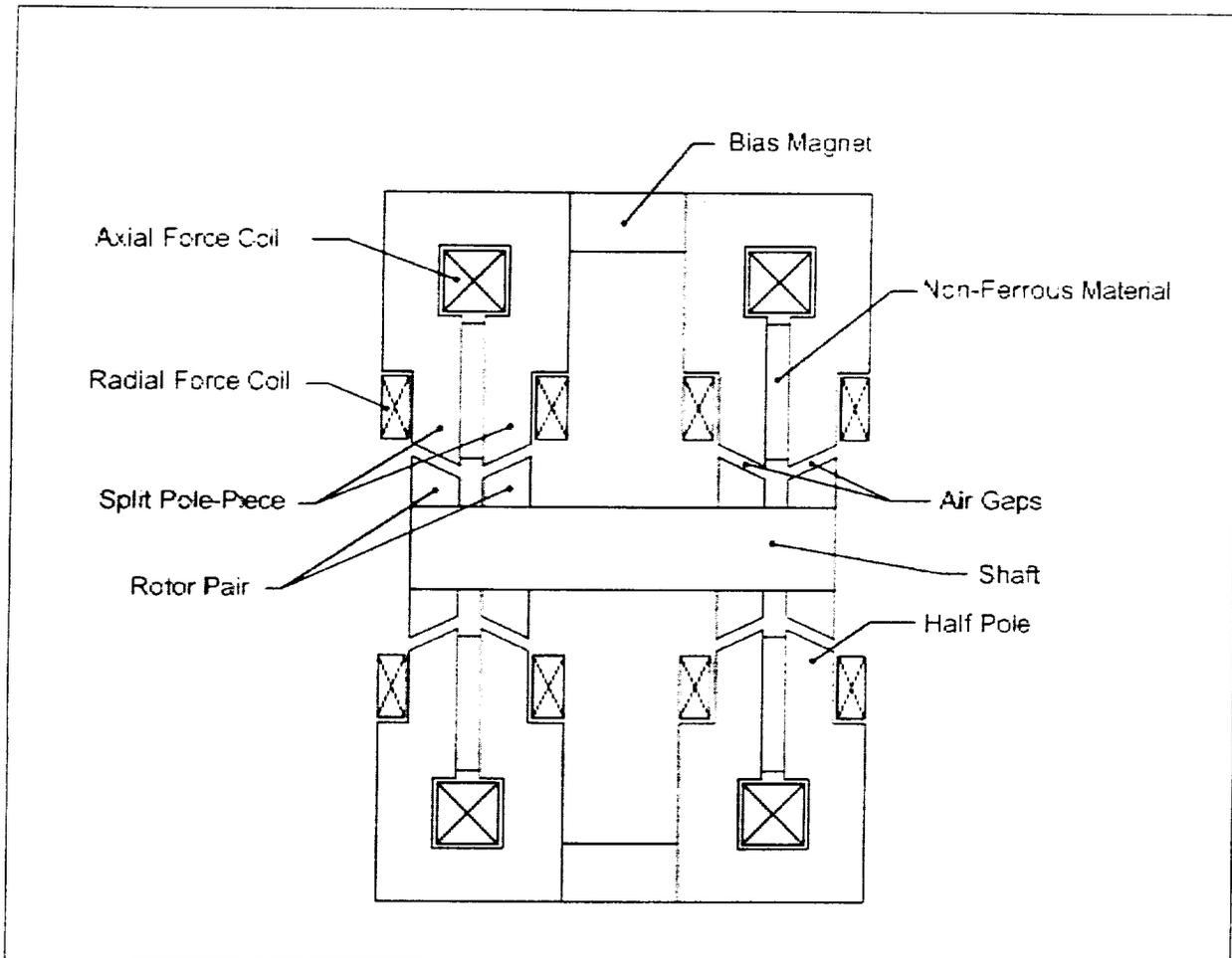


FIGURE 1: Cross-section of Radial and Axial Magnetic Bearing

This study prompted the development of a new and novel radial and axial magnetic bearing design. In order to minimize rotational losses, the new design was to be of homopolar configuration and take best advantage of lamination orientation. To minimize resistive losses, the new design had to have a configuration that did not require control fluxes to flow through bias magnets which greatly increases magnetic circuit reluctance. Since axial force capability needed to be comparable to radial force capability, the new design had to be able to achieve this with minimal coupling effects between axial and radial axes. It was also deemed desirable to maintain separate axial inputs and avoid a design that requires mixing of control signals which leads to a more complex control electronics scheme.

DESCRIPTION

A new configuration magnetic bearing was designed to have the desired characteristics previously stated. A

patent application for this design has been filed. The cross-section in Figure 1 resembles a radial homopolar magnetic bearing with a few obvious differences. A non-ferrous material separates pole-pieces in two parts called "split pole-pieces." The split pole-pieces are composed of half-poles that are tapered mirror images of one another. Radial force coils are wound around each split pole-piece. Corresponding conical rotor pairs to match the split pole-pieces are mounted on a shaft. An axial force coil is located around the outer diameter of each set of split pole-pieces within each stator. Bias flux distributes itself uniformly throughout the split pole-pieces so that flux feed radially into the left rotor pair of Figure 1, and radially out of the right rotor pair. Assuming that the shaft is centered both axially and radially, the force vectors of each half rotor, due to the bias flux, will cancel such that there will be no net axial or radial forces acting upon the shaft. However, if the shaft is displaced radially, there will be a non-uniform distribution of bias flux resulting in an attraction to the side with the smaller air-gap.

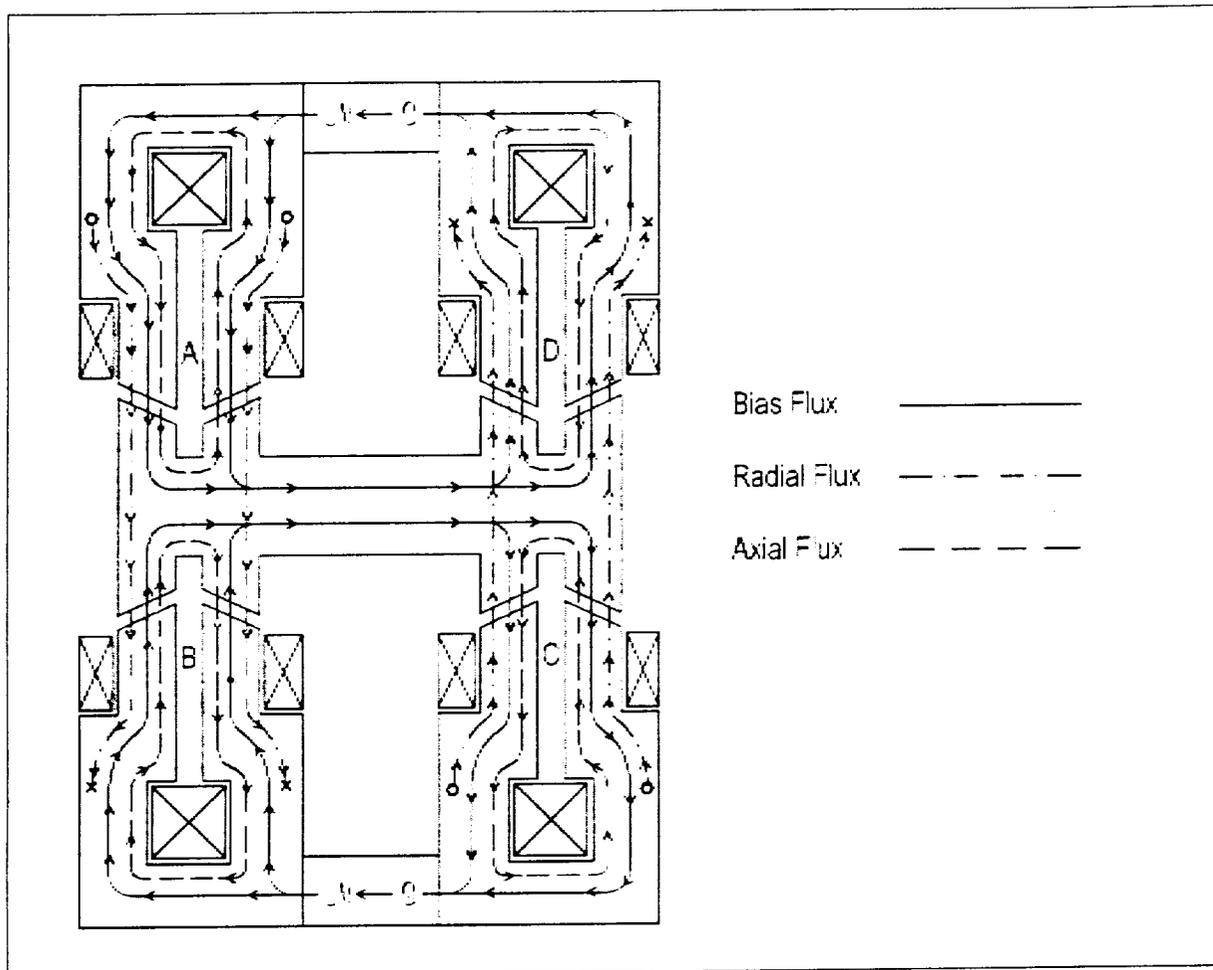


FIGURE 2: Magnetic Flux Paths

Likewise, if the shaft is displaced axially, there will be an attractive force acting on the shaft in the direction toward the smaller air-gap. This unstable condition is typical of magnetic bearing behavior. The unstable behavior of this new configuration magnetic bearing would be similar to that of radial force only and axial force only magnetic bearings mounted in tandem. However, there are coupling considerations to consider when the axis is off-center. Analysis will be shown with regards to cross-coupling later in this paper.

Radial control fluxes are as follows referring to Figure 2. Radial control coils are energized such that radial control flux flows into a rotor pair from split pole-piece A and out of the rotor pair into split-pole piece B. Radial control flux also flows from split pole-piece C into its rotor pair and out of that rotor pair into split pole-piece D. The control fluxes add to the bias fluxes in the upper air-gaps and subtract from the bias fluxes in the lower air-gaps resulting in a force on the shaft that would move it toward the top of the page.

Before discussing the flux paths resulting from energizing the axial coils, it should be noted that the radial behavior of this new design is virtually identical to that of a radial force only homopolar magnetic bearing. The tapered split pole-pair resultant force vector will be in the radial direction when radial force coils are energized because of symmetrical fluxes. If the tapers were eliminated from the poles, and the axial coils ignored, the magnetic bearing would become a radial force only homopolar magnetic bearing. The flux paths and theory of operation would be identical. When axial force coils are energized, a flux flows from the left side of split pole-pairs A and B radially into the left side of the rotor pair. Flux flows radially outward from the right side of the rotor pair into the right sides of split pole-pairs A and B. Note that this flux adds to the bias flux in the left air-gap and subtracts the bias flux in the right air-gap. This results in a force that would move the shaft axially to the right. Looking at split pole-pairs C and D, the flux also adds to the bias flux on the left side of the split pole-pairs, and subtracts

from the bias flux on the right, also resulting in a force on the shaft which will move it to the right.

MAGNETIC CIRCUIT ANALYSIS

Once the dimensions were selected, a quasi-static nonlinear analysis was done to predict the magnetic bearing forces generated by the bias magnets and control currents. These forces were calculated over many iterations of shaft positions in all three translational degrees of freedom. The analysis used a magnetic circuit equivalent. Voltage sources replaced MMF sources (control coils), current sources replaced flux sources (permanent magnets), and resistors replaced reluctances (air gaps). The reluctance of the iron was assumed to be zero, and the nonlinear characteristics of the materials were not included in this analysis. The air gap reluctances were calculated as functions of the shaft position using trigonometric relations such as:

$$R = (g - x \cdot \cos(\theta) - z \cdot \sin(\theta)) / (\mu \cdot A) \quad (1)$$

where:

- R = reluctance of the air gap
- g = nominal air gap (shaft centered)
- x = radial displacement from nominal
- z = axial displacement from nominal
- A = pole piece cross sectional area
- μ = permeability of free space
- θ = slant angle of the pole pieces

The control current MMF sources were modeled as sources having the voltage:

$$F = N \cdot I \quad (2)$$

where:

- F = mmf in amp-turns
- N = number of turns on the control coil
- I = control current in amps

The 3-dimensional bearing model became a simple electrical circuit model. By using the principles of superposition and Kirchoff's Current Laws, a software routine was written to vary the gap in both radial directions and the axial direction and to solve for the flux in each gap at each position. Once the fluxes were known in each gap, then the corresponding radial and axial forces were calculated using:

$$Force = (\phi \cdot R) / (2 \cdot \mu \cdot A) \quad (3)$$

where:

- R = reluctance of the air gap
- ϕ = flux in the air gap
- A = pole piece cross sectional area
- μ = permeability of free space

By solving the radial and axial forces for many permutations of position, mesh plots were created to provide insight into the cross-coupling between the axes. The forces were separated into bias forces, which are the natural negative spring constants of the magnetic bearing, and the net force constant of each of the bearing coils. Three dimensional curve fits of the data provided nonlinear equations relating bearing forces to control current and shaft position. These parameters were later used to create a dynamic model of the bearing for controls design purposes. The results of the analysis showed that the bearing is very linear and well decoupled near the nominal gap. As the shaft is displaced, both the bearing bias and control forces become nonlinear with shaft position.

Several examples of force characteristics for forces in the axial axis (Z) are shown in this paper. Figure 3 shows the mesh plot of how the bias force in Z (axial) varies over position changes in Z and X (radial). Ideally, this would be a flat plane with a slope in the Z-axis, but flat in the X-axis. This is the case towards the center of the plot. However, the surface becomes non-ideal at extreme values of X and Z. (Note that the scale shows ± 0.25 mm of displacement in Z and ± 0.13 mm in X. This is due to the slant angle of about 30° that allows more motion in Z than X before closing the gap.) Increased nonlinearity at the positional extremes indicates that the natural bearing spring constant will be significantly different during the initial levitation than when it is centered in the gap. There is a nearly identical relationship for the bias forces in X versus X and Z displacements.

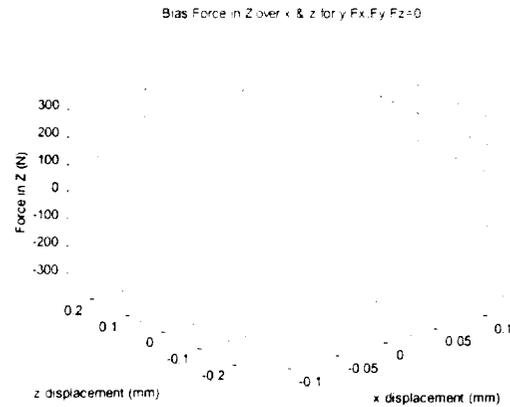


FIGURE 3: Variance of Bias Flux Forces with Position

Similarly, the bearing force constant varies with position. Figure 4 shows the relation between input current to the Z axis and force in the Z axis versus displacement in X. This shows that the bearing is well decoupled here in that the peak force values (in Z) only change about 8% across the range of motion in X.

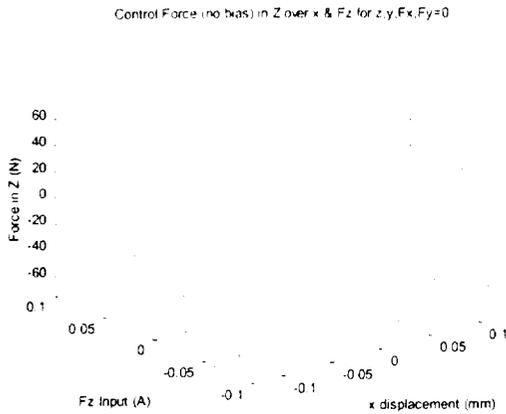


FIGURE 4: Z Control Force versus X Position

Finally, the analysis shows the ability to decouple the axes. Figure 5 shows a quite nonlinear mesh indicating that for a given control current in the X axis (F_x) that the bearing actually generates a force in Z that is highly dependent on the Z location. However, the peak value of this nonideal force is about 20N for a maximum input current in X and maximum displacement in Z. For comparison, force capability of the bearing to current in Z is over 60N. Again, once the bearing is levitated near the nominal gap, this coupling drops to less than 2N and is quite linear. This effect can be included into the controls design for a MIMO system.

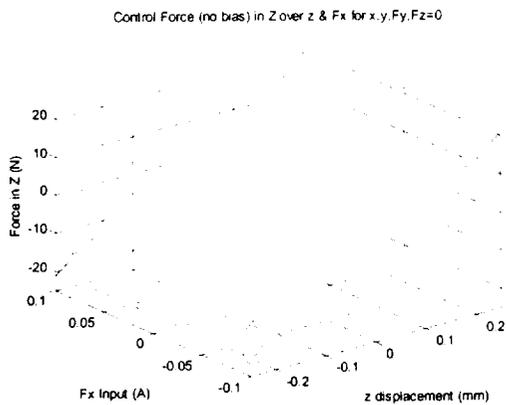


FIGURE 5: Variance of Forces in Z with X Input

The magnetic circuit model analysis shows the design to have excellent linearity and axis decoupling when the shaft is nearly centered. At the extreme ends of the shaft displacement, the bearing exhibits some nonlinear behavior, as do almost all other magnetic bearing designs.

CONCLUSION

A new and novel magnetic bearing configuration has been designed to provide forces in both radial and axial directions in a single unit. Combining radial and axial functions into a single unit improves rotordynamics for high-speed machines as well as simplifies the design of a mechanism that utilizes magnetic bearings that are active in all axes. This new design has eliminated some serious drawbacks that have made previous radial and axial magnetic bearing designs to be less than desirable. It can also provide force in both radial and axial directions with minimal resistive losses and maximal rotational efficiency. Coupling between X and Z axes may require limitation of the pole taper angle, limitation of Z displacement, or controller decoupling. An energy storage flywheel is an ideal candidate for application of this new magnetic bearing configuration since improvements in rotational efficiency and rotordynamics are crucial for the practical application of this technology. This new configuration magnetic bearing can be used for all applications where a fully active magnetic bearing system is desired along with highest efficiencies.

REFERENCES

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